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DISTORTION OF GAS THRUST BEARING DUE TO VISCOUS SHEAR

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DISTORTION OF GAS THRUST BEARING DUE TO VISCOUS SHEAR

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TABLE OF CONTENTS

INTRODUCTION

While gas bearings have low friction losses in comparison with liquid bearings, they are far from being "frictionless", and in some cases the losses are larger than those for rolling element bearings. In fact, the pressure generation results from viscous shear. The heat generated by viscous shear can be sufficiently large to cause appreciable thermal distortion which in turn affects the pressure generation and the load carrying capacity. The effect of thermal distortion must be considered in design especially where high speeds and/or large radius ratio bearings are used. This is especially important in instruments (e.g. gyros, accelerometers), cryogenic turbo-expanders, low power level Brayton cycle turbo-machinery, etc. In these systems the friction loss detracts from overall efficiency. As an example, at cryogenic helium temperatures a watt loss in the cold turbo-expander represents 100 - 1000 watts of additional compressor power required. In these applications it is therefore important to design the smallest possible bearing which will support a given load with a prescribed film thickness. Any distortion is undesirable because it reduces the load carrying capacity and increases the power losses. This paper presents an analysis of thermal distortion of a gas lubricated spiral-grooved thrust bearing resulting from viscous shear.

The distortion caused by viscous shear is a function of the bearing gap, fluid viscosity, and rotor speed; the degree of distortion depends on how the heat is removed through the structure of the bearing and the amount of dimensional change induced by temperature variations. Thus a complete analysis of this problem would include the construction of a detailed thermal map of the bearing parts. However, an appreciation of the gross phenomenon of thermal distortion can be gained through a much simplified analysis. The present work deals with the latter approach concerning typical spiral-grooved bearings.

The spiral-grooved thrust bearing or Whipple plate has been chosen for the analysis because it is most widely used in gas bearing applications. This is because of its simplicity and because it possesses high load carrying capacity. For instance, at conditions typical of gas bearing gyroscopes the optimized spiral-grooved thrust bearing has about twice the load-carrying capacity of an optimized stepped thrust bearing according to theory (Ref. 1, 2 and 3).

The theoretical advantage of spiral-grooved thrust bearings is often not fully realized in practice. Fabrication difficulty has been regarded as the prime factor degrading the actual bearing performance. Recent advances in manufacturing techniques have all but completely removed this problem. There is also an inherent reason why the theoretical performance presently available cannot be fully realized in practice. This is because an infinite number of grooves (each of infinitesimal width) is assumed in the theory, whereas in practice a finite number of grooves are present. This assumption reduces the load capacity in two ways:

1) Along the ambient edge, a constant pressure is maintained, therefore end leakage exists. This effect has been found to be relatively insignificant when there are more than fifteen grooves in the bearing (Ref. 2).

2) The bearing load capacity for spiral grooved thrust bearing increases linearly with speed until density variation within a groove becomes sufficiently large that further speed increase would no longer yield a corresponding increase in load capacity. This latter effect may be regarded as a compressibility threshold which can be estimated according to the infinite groove theory and can be avoided if the number of grooves can be made larger than a lower limit, which is approximately proportional to the compressibility number of the bearing (Ref. 4).

With the refined manufacturing techniques and analysis there still have been many examples where theory and practice deviated appreciably. This discrepancy can be largely attributed to distortion. Distortion can result from several sources such as material stability, internal stresses, pressure and thermal distortion. The effect of thermal distortion as shown in this paper, can have a very marked influence on load carrying capacity and therefore must be considered in design.

GENERAL EFFECTS OF UNEVEN HEATING

When heat flows through a material, due to its thermal resistivity, there is a temperature gradient against the direction of heat flux according to Fourier's law of conductivity. Because the density of the material usually depends on the temperature, the presence of temperature gradient would affect both size and shape of the body and would also create internal stresses. Given the temperature field, the structural constraints, and the pertinent properties of the material, all these effects can be determined in detail from thermoelastic analysis. Clearly, like all precision devices, the performance of a gas bearing can be immensely influenced by the temperature effects on the size and the shape of the bearing parts.

The size effect is primarily dependent on the temperature level. If the materials of the journal and bearing have different coefficients of thermal expansion, then the bearing gap would vary with temperature. If a reference mean radial gap is measured at a reference temperature, the mean radial gap would deviate from the reference value by an amount proportional to the product of the journal radius and the difference of the coefficients of thermal expansion and the temperature difference from the reference temperature. If the coefficient of thermal expansion of the bearing is larger, a rise of temperature above the reference value would increase the radial gap. A similar situation exists with double-acting thrust bearings.

To better understand the type of shape variation or distortion of bearing as may be caused by a non-uniform temperature field, consider a circular cylinder shown in Fig. 1. If the temperature gradient is purely radial, the cross-section would become tapered due to lengthwise lineal expansion. The degree of taper is proportional to the length of cylinder. Also hoop stress would be induced such that Poisson's ratio effect would further increase the amount of taper in proportion to the cylinder length. Except for localized three dimensional effects near the two ends, the cylindrical surfaces would remain cylindrical. The main distortion is the non-flatness of the ends. The situation is illustrated in Fig. 1 (a). This problem is important if the thrust bearing is at the end of a heat generating body.

If the temperature gradient is transverse, the cylinder would become bent as shown in Fig. 1 (b). This situation can lead to serious misalignment difficulties for both the cylindrical and the end surfaces. This problem can be alleviated by avoiding geometrical dissymmetry and by appropriate thermal shielding from external heat sources.

If the temperature gradient is longitudinal, the cylindrical surfaces would become tapered or conical, and the end faces would become spherical. Thus this form of distortion would impair the performance of both journal and thrust bearings. In the case of the thrust bearing, the cause for this type of distortion is built-in because there is ample heat dissipation in the bearing film which would create an axial temperature gradient. The present work is concerned with the latter problem with specific reference to spiral-grooved thrust bearings.

There are other factors in thermal distortions, such as the bi-metal phenomenon and structural constraints. The significance of each of these factors vary according to design. They are neglected in the present work.

Distortion of the Thrust Surface due to Self-Heating

A first approximation of the distortion of the thrust bearing due to viscous shear in the bearing gap can be obtained by assuming that heat flow through the bearing is uniform and is purely axial. Further, for simplicity, the bearing plate is assumed to be a hollow disk. Let q'' be the heat flux, then the axial temperature gradient is

$$\frac{\partial T}{\partial z} = \frac{q''}{\kappa} \quad \dots\dots\dots\dots\dots\dots\dots\dots \quad (1)$$

where κ is the coefficient of thermal conductivity; and the isotherms are planes parallel to the bearing surface as shown in Fig. 2. The disk is presumed to be flat if maintained at a uniform temperature.

Consider an infinitesimal radial line measured to be δr at a reference temperature T_o, then when this line is at $T(z)$ its length is

$$\delta r \left\{ 1 + \alpha \left[T(z) - T_o \right] \right\}$$

where α is the coefficient of lineal thermal expansion. A similar infinitesimal radial line which was of the same length at the reference temperature but situated at $z + dz$ would now measure to be

$$\delta r \{1 + \alpha \; [T (z + dz) - T_o]\}$$

Under the influence of the axial temperature gradient, these lines deform into circular arcs; and surfaces originally normal to the undeformed axis now become spherical surfaces. Assume unrestricted deformation, then the radius of curvature, R, can be deduced according to the diagram in Fig. 3, in terms of the following relations:

$$\frac{\delta r \; \{1 + \alpha \; (T - T_o)\}}{R} = \frac{\delta r \alpha \; [T \; (z + dz) - T \; (z)]}{dz}$$

Or,

$$R \frac{\alpha q''}{\kappa} = 1 + \alpha \; (T - T_o) \quad \ldots\ldots\ldots\ldots \quad (2)$$

Because α is of the order of $10^{-5}/^{o}F$, $\alpha(T-T_o)$ is typically a very small number. Therefore, as a good approximation,

$$R \frac{\alpha q''}{\kappa} \approx 1^{*} \quad \ldots\ldots\ldots\ldots \quad (3)$$

It is interesting to note that R is not explicitely dependent on either the thrust disk thickness or the temperature level. α and κ are material properties, and q'' depends on the rotor speed, radius, gap, lubricant viscosity, and the detail gap geometry.

Consider now the total amount of heat generated in the bearing film is divided between the two parts of the bearing. Let subscripts "1" and "2" designate various quantities associated with the two bearing parts, and let h_o be the gap along the center line, then the gap at any radius is

$$h = h_o + R_1 \; \{ 1 - \cos \; (\sin^{-1} \frac{r}{R_1})\}$$
$$+ R_2 \; \{ 1 - \cos \; (\sin^{-1} \frac{r}{R_2})\} \quad \ldots\ldots\ldots\ldots \quad (4)$$

If $\frac{r}{R_1}$ and $\frac{r}{R_2}$ are small, Eq. (4) can be approximated by

$$h \approx h_o + \frac{r^2}{2} \{\frac{1}{R_1} + \frac{1}{R_2} \} \quad \ldots\ldots\ldots\ldots \quad (5)$$

Substituting Eq. (3) into Eq. (5), one obtains

$$h \approx h_o + \frac{r^2}{2} \left\{ \frac{\alpha_1 q_1''}{\kappa_1} + \frac{\alpha_2 q_2''}{\kappa_2} \right\} \quad \ldots\ldots\ldots\ldots \quad (6)$$

Note that if α/κ is same for both parts of the bearing, then the bearing film shape

---

* This result was previously given in Ref. 5

becomes only a function of the total amount of heat generated in the bearing film, but is independent of how it is divided between the two parts. In subsequent work, the subscripts will be dropped and two terms on the right hand side of Eq. (6) will be combined such that

$$h \approx h_o + \frac{r^2}{2}\frac{1}{R} \quad \ldots\ldots \quad (7)$$

In the event that q'' is unevenly divided and the material properties are different, it is understood that Eqs. (3) and (7) can still be used provided we define

$$\frac{\alpha}{\kappa} = \frac{1}{q''}\left[\frac{\alpha_1 q_1''}{\kappa_2} + \frac{\alpha_2 q_2''}{\kappa_2}\right] \quad \ldots\ldots \quad (8)$$

Heat Generated in Deformed Bearing Gap

The heat generated in the bearing film can be equated to the power required by the friction torque. The friction torque is the integrated moment of shear stress over the entire bearing surface. The shear stress, in turn, depends not only on the relative sliding motion between the opposing surfaces but also on the circumferential pressure gradient in the bearing film. Thus the friction torque would depend on the detail geometrical parameters of the bearing gap; in the case of a spiral-grooved thrust bearing, they include radius ratio, angle of inclination, groove depth, groove width ratio, and groove length. In Ref. 2, the friction troque for typical spiral grooved thrust bearings of parallel surfaces has been calculated, it was found to be about 70% of that for plain parallel disks rotating at the same speed at the same minimum gap. The reduction in friction torque is primarily caused by the shallow grooves which are on the outer portion of the bearing. If they were on the inner radius of the bearing, the torque would approach that of plain disks. Thus it would be quite satisfactory for the present purpose to calculate q'' according to a pure sliding motion between smooth surfaces separated by the gap as given by Eq. (7).

The shear stress of an area element $rdrd\theta$, is

$$\tau = \frac{\mu\omega r}{h}$$

where μ is the viscosity of the fluid and ω is the angular speed. The mechanical power required to overcome the friction per unit area is:

$$\tau\omega r = \frac{\mu\omega^2 r^2}{h}$$

The total frictional power for the entire disk is

$$H = \int_0^{2\pi} \int_{r_i}^{r_o} \frac{\mu\omega^2 r^3}{h} \, drd\theta$$

$$= 2\pi\mu\omega^2 \int_{r_i}^{r_o} \frac{r^3 \, dr}{h_o + \frac{r^2}{2R}}$$

The integration can be readily performed after expanding the integral by the method of partial fractions. The result is

$$H = 2\pi\mu\omega^2 R \left\{ (r_o^{\,2} - r_i^{\,2}) - 2\, h_o R \, \ell n \left[\frac{1 + \frac{r_o^{\,2}}{2h_o R}}{1 + \frac{r_i^{\,2}}{2h_o R}} \right] \right\} \quad \ldots\ldots\ldots\ldots \quad (9)$$

In a gas bearing, convective heat transfer in the film is negligible, therefore all the heat generated by the frictional power goes through the bearing surfaces, and the total heat flux per unit area is

$$q'' = \frac{H}{J\,\pi\,(r_o^{\,2} - r_i^{\,2})} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

q'' can be eliminated between Eqs. (3) and (10), also, making use of Eq. (9), one finds

$$1 = \frac{2\mu\omega^2 R^2}{J} \, \frac{\alpha}{\kappa} \left\{ 1 - \frac{2\, h_o R}{(r_o^{\,2} - r_i^{\,2})} \, \ell n \left[\frac{1 + \frac{r_o^{\,2}}{2h_o R}}{1 + \frac{r_i^{\,2}}{2h_o R}} \right] \right\} \quad \ldots\ldots\ldots\ldots \quad (11)$$

It is useful to define the following dimensionless groupings -

Distortion Parameter: $\Delta = \left(\frac{2J}{\mu\omega^2 r_o^2}\right)\left(\frac{\kappa}{\alpha}\right)\left(\frac{h_o}{r_o}\right)^2$

Curvature Index: $K = \frac{r_o^2}{2h_o R}$ . . . . (12)

Radius Ratio: $\mathbb{R} = r_i \;/\; r_o$

The distortion parameter represents the relation between frictional heat and the radius of curvature due to thermal distortion. The curvature index is actually the ratio of the amount of non-flatness at the bearing rim to the separation of the bearing surfaces along the axis of rotation (see Fig. 4). Using the new definitions Eq. (11) can be written as:

$$K^2\Delta = \frac{1}{(1-\mathbb{R}^2)\,K}\,\ell n\left[\frac{1+K}{1+\mathbb{R}^2 K}\right] \quad \text{. (13)}$$

Eq. (13) is graphically shown in Fig. 5.

It is of interest to examine the above equation for extreme values of K. If the amount of distortion is very small, K << 1, then Eq. (13) reduces to

$$\lim_{K \ll 1} \Delta = \frac{1}{2}\,(1+\mathbb{R}^2)\;/\;K \quad \text{. . . . . (14)}$$

If the amount of distortion is very large, $\mathbb{R}^2 K \gg 1$, then Eq. (13) is reduced to

$$\lim_{\mathbb{R}^2 K \gg 1} \Delta = \left\{1 - \frac{1}{(1-\mathbb{R}^2)\,K}\,\ell n\,\frac{1}{\mathbb{R}^2}\right\} / \;K^2 \quad \text{. . . . . (15)}$$

Load Capacity of Distorted Spiral-Grooved Thrust Bearings

Analysis of the spiral-grooved, gas lubricated thrust bearing was given in Ref. (4). Geometry of such a bearing is illustrated in Fig. 6. Allowing for gap variation due to thermal distortion, the fluid film pressure is governed by the following equations:

$$-\,f_1\,\zeta\,P\,\frac{dP}{d\zeta} + \Lambda f_2\,\zeta^2 P = 2\,\Lambda_f \quad \text{. . . . . (16)}$$

where

$\zeta = r/r_o$,

$P = p/p_a$,

$\Lambda = 6\mu\omega\, r_o^2/\,(p_a\, h_o^2)$

Λ_f is an integration constant.

f_1 and f_2 are coefficients determined by the gap geometry. In the grooved region,

$$f_1 = \left(\frac{h}{h_o} \right)^3 \frac{\Gamma^3 + a(1-a)\ \sin^2\ (\Gamma^3-1)^2}{a + (1-a)\ \Gamma^3} \quad \ldots\ldots\ldots\ldots \quad (17)$$

$$f_2 = \left(\frac{h}{h_o} \right) \frac{a(1-a)(\Gamma^3-1)(\Gamma-1)\ \sin\beta\ \cos\beta}{a + (1-a)\Gamma^3}$$

where,

$$\Gamma = 1 + (\Gamma_o - 1)\ h_o/h$$

$$\Gamma_o = 1 + \delta/h_o,$$

$$a = a_g/a_r;$$

In the seal region,

$$f_1 = \left(\frac{h}{h_o} \right)^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (18)$$

$$f_2 = 0$$

There are three boundary conditions to be met.
They are:

$P(\zeta_i) = 1$

$P(1) = 1$

$P(\zeta)$ is matched between the grooved and seal regions.

The only differences between above equations from those employed in Ref. 4 are due to gap variation and are contained in the coefficients f_1 and f_2.

Eq. (16) is solved on a digital computer using the Runge-Kutta method (Ref. 6). Λ_f is determined by successive approximation to permit matching of $P(\zeta_m)$ between the two regions. Upon finding P, the load capacity, W, is readily found by appropriate numerical quadrature. In dimensionless form, this is

$$\frac{W}{\pi r_o^2\ p_a} = 2 \int_{\zeta_i}^{1} (P-1)\ \zeta d\zeta \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (20)$$

Allowing for thermal distortion, the load capacity of spiral-grooved thrust bearings now depends on the curvature index K in addition to the other parameters which include Λ, $\mathcal{R}$, β, Γ_o, a and ζ_m. However, K is itself a consequence of thermal distortion and is not a convenient design parameter. Fortunately, K is related to $\mathcal{R}$ and Δ according to Eq. (13) and can be read from Fig. 5. Thus Δ, instead of K, is the additional input parameter available to the designer. Upon reading K from Fig. 5, one can then proceed to calculate the load capacity of the distorted thrust bearing.

Numerical Example

Computation of the load capacity of spiral grooved thrust bearings with thermal distortion effects according to the procedure outlined above is a straight forward matter. However, because of the large number of design parameters involved, it is impractical to consider every aspect of the problem. In the following, a typical example will be considered in order to bring out significant points of the present analysis.

The results of the example are shown graphically in Figs. 7 and 8. Physical and geometrical variables of the bearing in concern are given in the legends of these figures. This bearing without distortion would be able to carry 92 lbs. with a design gap of 0.8 mil. Once the bearing material is chosen the value of α/κ is fixed. Then for each value of h_o, there would be corresponding values for Δh and W. Fig. 7 contains essentially the same information as Fig. 5, except various quantities are given in appropriate units. Fig. 8 shows that considerable loss of load capacity or minimum gap can occur for $\alpha/\kappa > 10^{-7}$ ft-hr/Btu. At $\alpha/\kappa = 2 \times 10^{-7}$ ft-hr/Btu, the load is down to 60 lbs at 0.8 mil gap, or the gap would be 0.62 mil in order to carry the design load of 92 lbs. Also shown in Fig. 8 are the ranges for α/κ for typical structural material, according to Ref. 7. It is noted that most common materials have α/κ in excess of 10^{-7} ft-hr/Btu.

Generalization

Clearly, loss of load capacity is directly related to K, which in turn, as seen from Fig. 5; is essentially a function of Δ. Since Δ contains other factors than α/κ, it is useful to examine what other factors enter into the picture. The definition for Δ can be rewritten as

$$\Delta = \frac{12J}{\Lambda} \frac{\kappa}{p_a r_o^2 \omega\alpha} \qquad (21)$$

Since the load capacity of spiral grooved thrust bearings is proportional to $\Lambda p_a r_o^2$, one can rewrite the above expression as

$$\Delta \sim \frac{1}{\omega W} \left(\frac{\kappa}{\alpha}\right)$$

From this one concludes that the importance of thermal distortion for a given material increases directly with the bearing speed and load. Actually, according to Refs. 2 and 4, for bearing optimized for load capacity, $W / (\Lambda p_a \pi r_o^2)$ varies with $\mathcal{R}$ as shown in Fig. 9. From Figs. 5, 7 and 8 it looks reasonable to expect thermal distortion to become significant for $\Delta < 1.0$ ($\Delta h \gtrsim 0.5\, h_o$); the realizable load capacity may be only a fraction of the design goal due to thermal distortion. On the other hand, for $\Delta > 3.0$($\Delta h \lesssim 0.2\, h_o$), the achievable load capacity would be about 75% of the design load. Making use of Eq. (21) and Fig. 9, $\frac{J}{\omega W} (\frac{K}{\alpha})$ can be related to $\mathcal{R}$ for $\Delta =$ 1.0, 3.0 as shown in Fig. 10. Below the lower curve, thermal distortion would be excessive and drastic measures must be taken to achieve the design goal. Above the upper curve, distortion is not serious. In between, there is a reasonable chance to achieve the design goal by adjusting the radius ratio $\mathcal{R}$ and selection of material. However, in selection of material, attention must be paid to other design considerations, proper choice of α for mating materials, bearing friction and wear characteristics, material stability etc. Additional thermal distortion may result due to radial and axial thermal gradients existing in the equipment. Symmetrical design often can reduce radial gradients; most machines however, will have axial gradients.

CONCLUSIONS

1. Friction heat generated in the fluid film can distort the thrust surface and cause considerable reduction in load carrying capacity.
2. In order to minimize the frictional power losses the bearing distortion must be minimized.
3. The degree of distortion does not directly depend on the temperature level.
4. The amount of distortion is enhanced by a large coefficient of thermal expansion and is reduced by a large coefficient of thermal conductivity.
5. Large radius ratio bearings and/or high speed rotors are more sensitive to this type of thermal distortion.
6. Analysis of a typical bearing design shows that the effects of thermal distortion can be significant for most structural materials.
7. Materials for minimum thermal distortion are often not satisfactory with respect to structural considerations and from the standpoint of friction and wear. Application of surface coatings can alleviate the problem of friction and wear and permit the use of materials chosen for minimum thermal distortion.

NOMENCLATURE

a	groove-ridge width ratio, a_g/a_r
a_g/a_r	width of groove and ridge respectively
f_1, f_2	coefficients in the differential eq. for the spiral-grooved thrust bearing, eqs. (17), (18).
h	local gap
h_o	gap measured along the axis of rotation
Δh	increase of gap at rim due to distortion
J	Joule constant, 778 ft-lb/Btu
K	curvature index, $\Delta h/h_o$
p	absolute pressure in the fluid-film
p_a	ambient pressure
P	non-dimensional fluid film pressure, p/p_a
q''	axial or longitudinal heat flux
r	radius measured on the thrust surface
r_i, r_o	inner and outer radii of thrust bearing
r_m	radius at the groove-seal boundary
R	radius of curvature of distorted thrust bearing surface
$\mathcal{R}$	radius ratio, r_i/r_o
T	temperature
T_o	reference temperature
H	mechanical power required to overcome the friction torque
W	thrust load, lbs.
z	axial or longitudinal coordinate
α	coefficient of lineal thermal expansion, ft/ft/^{o}F
β	spiral angle of grooves, measured from the circumference
Γ	local groove-ridge gap ratio
Γ_o	nominal groove-ridge gap ratio
δ	groove depth
ζ, ζ_i, etc.	r/r_o, r_i/r_o, etc.
κ	coefficient of thermal conductivity, Btu/ft/^{o}F/sec. for Btu/ft/^{o}F/hr. as noted.

Λ	compressibility number
Λ_f	constant of integration
μ	viscosity of lubricant
Δ	distortion parameter, $12J\kappa/(\Lambda p_a r_o^2 \omega\alpha)$
ω	angular speed, radians/second

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LIST OF FIGURES

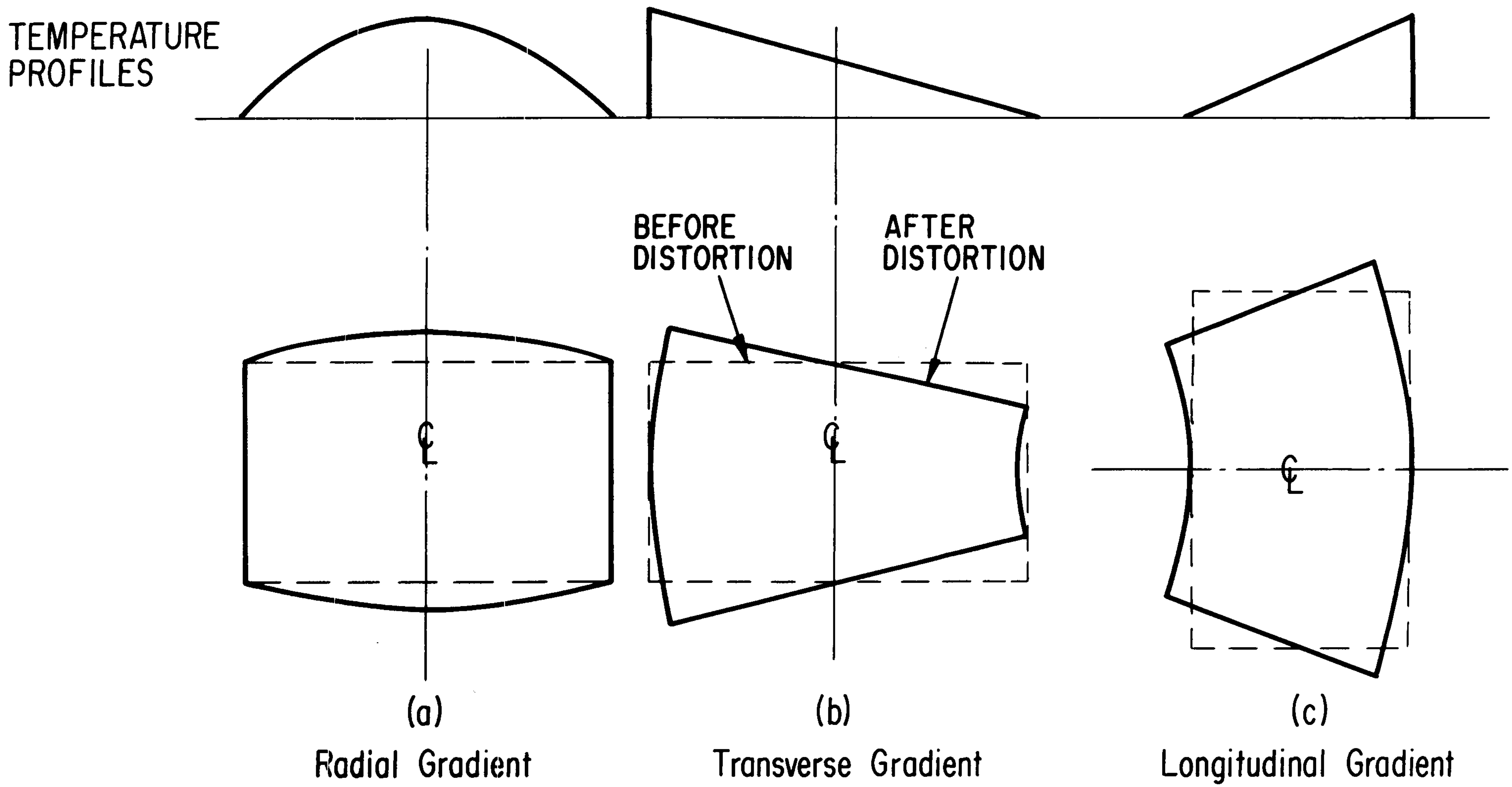

Fig. 1 Thermal Distortions of Cylinder.

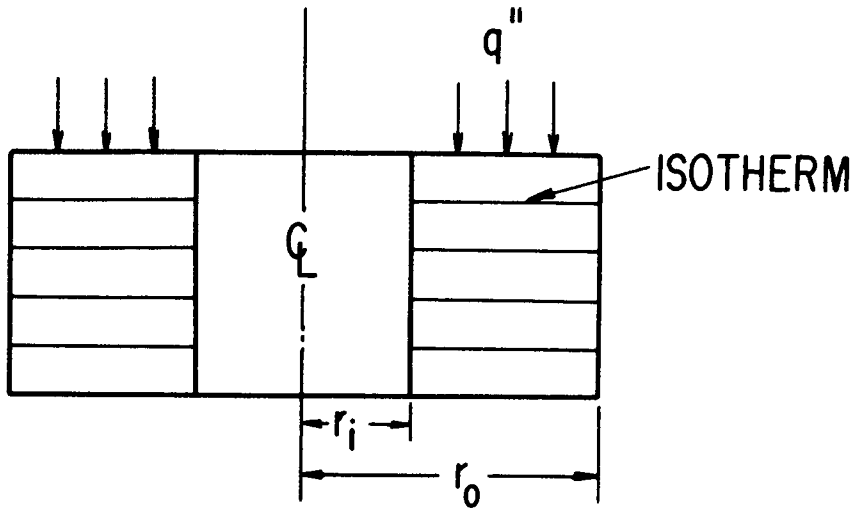

Fig. 2 Simplified Thermal Map in Thrust Bearing Due to Friction Heating.

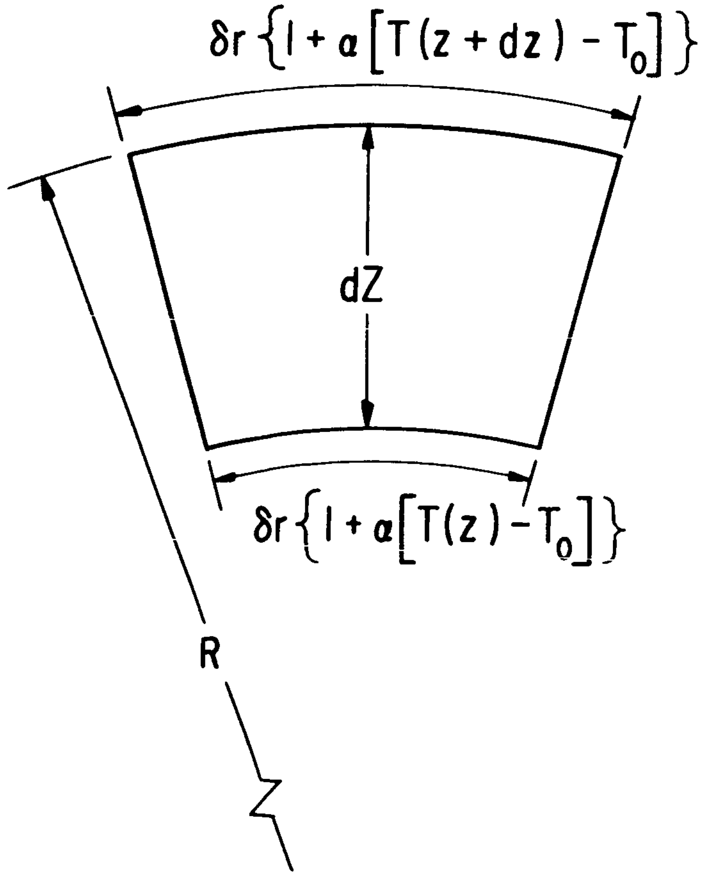

Fig. 3 Infinitesimal Distortion of an Element in Uiformal Thermal Gradient.

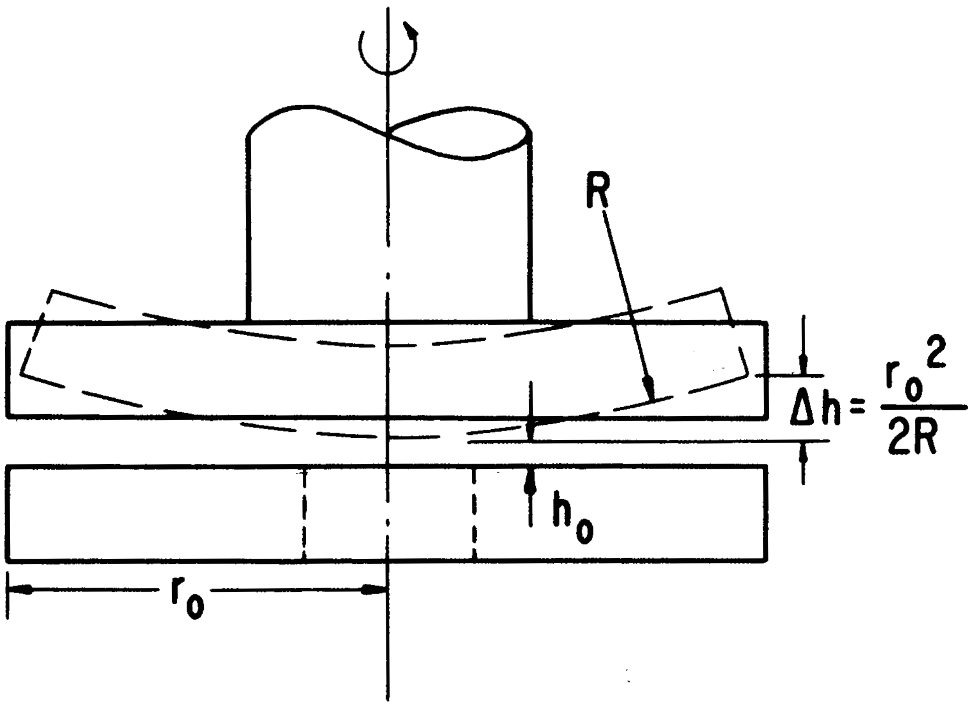

Fig. 4 Distortion of Thurst Plates Due to Heat Generation in the Fluid Film.

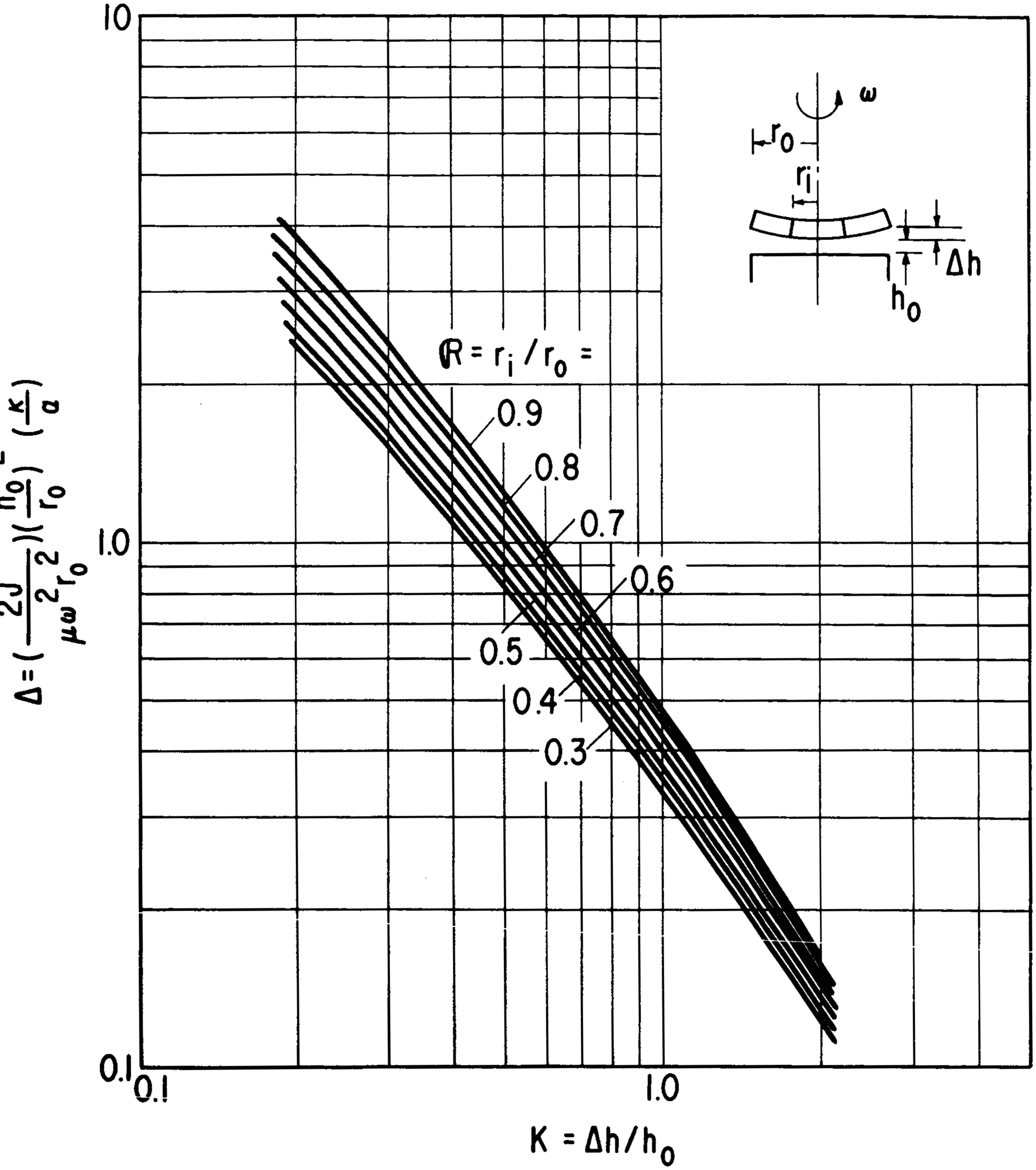

Fig. 5 Distortion of Annular Disk Due to Self-Heating.

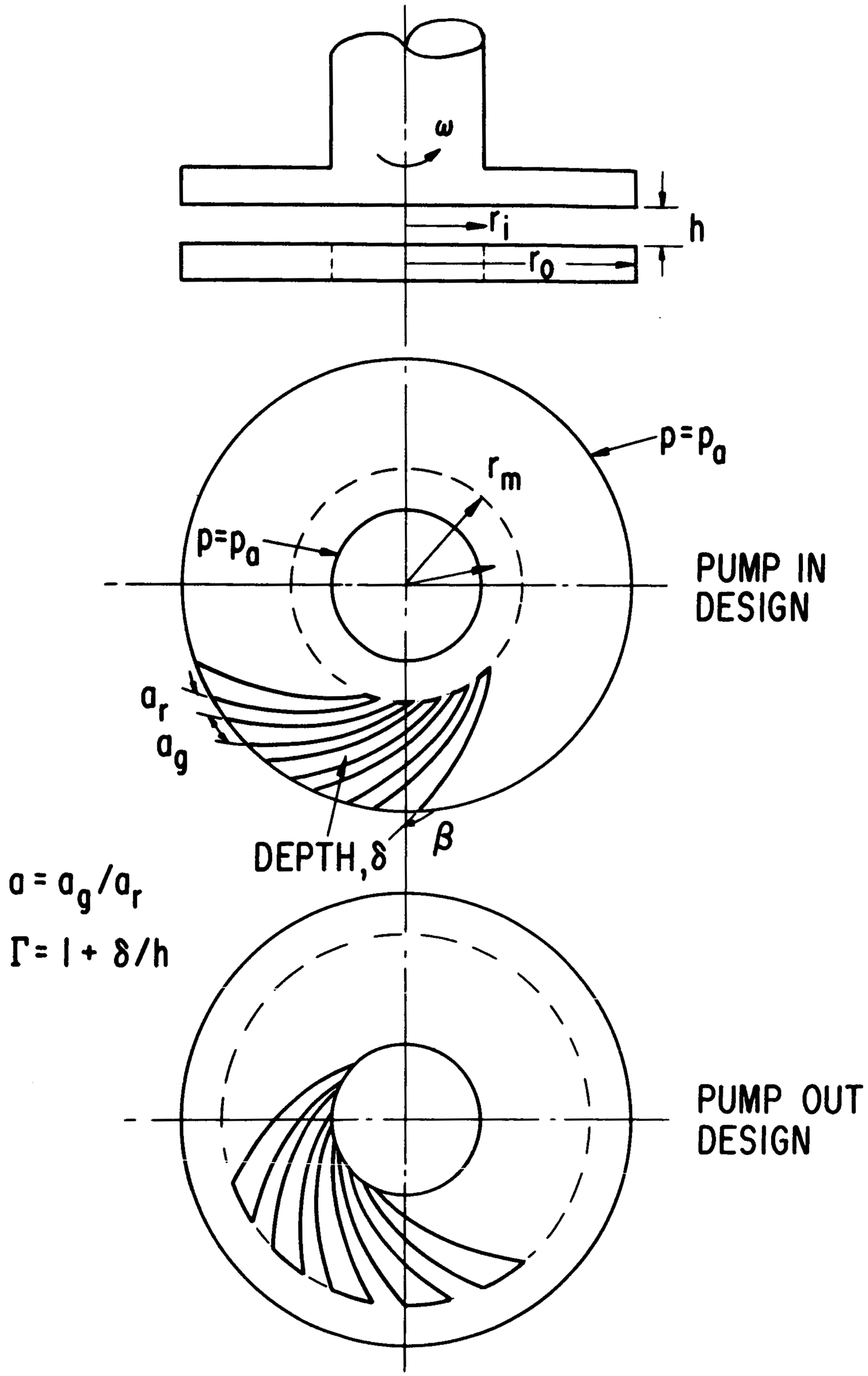

Fig. 6 Schematic of Spiral-Grooved Thrust Bearing.

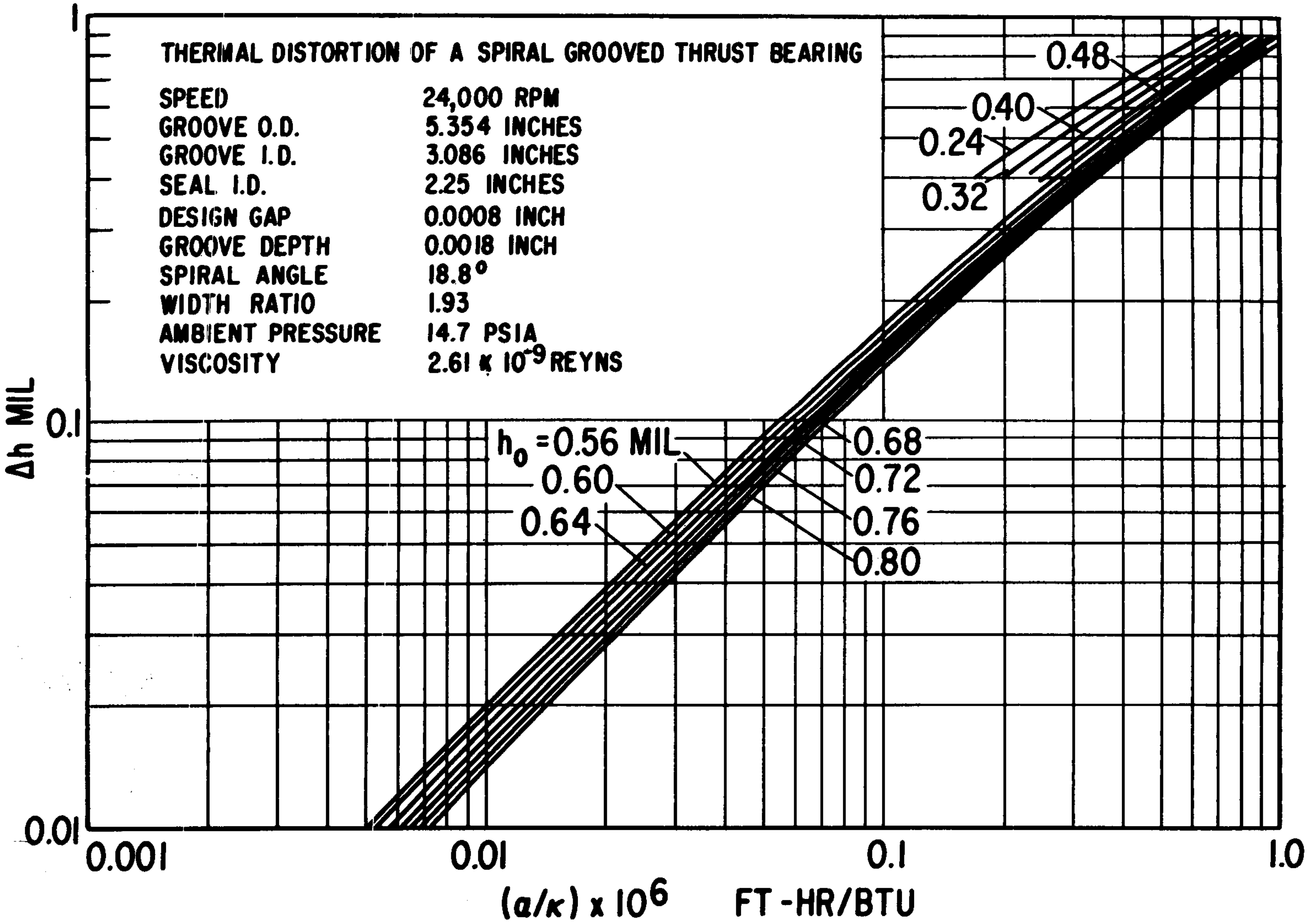

Fig. 7 Thermal Distortion of a Spiral-Grooved Thrust Bearing.

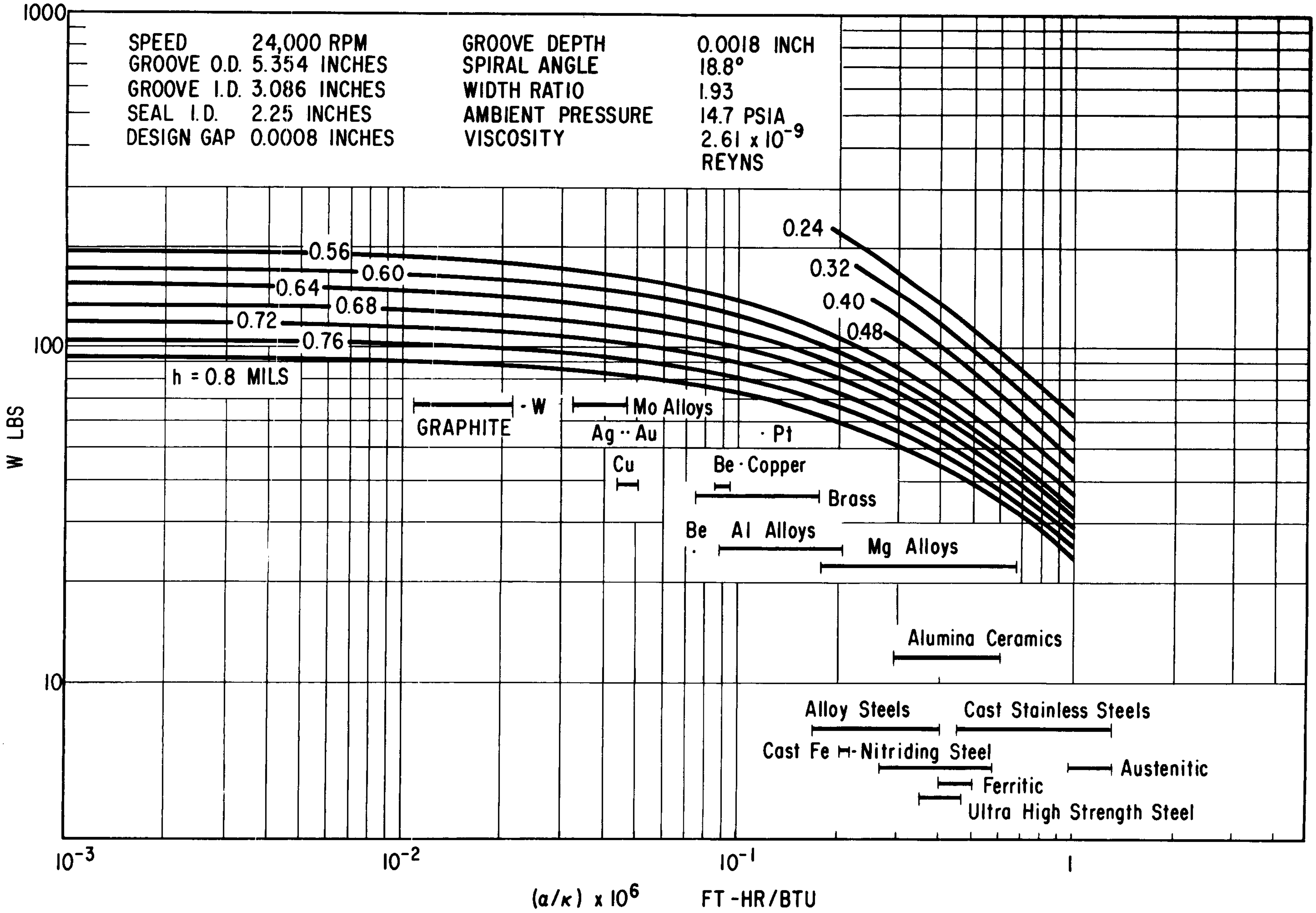

Fig. 8 Influence of $(\frac{\alpha}{\kappa})$ on Thrust Bearing Load Capacity.

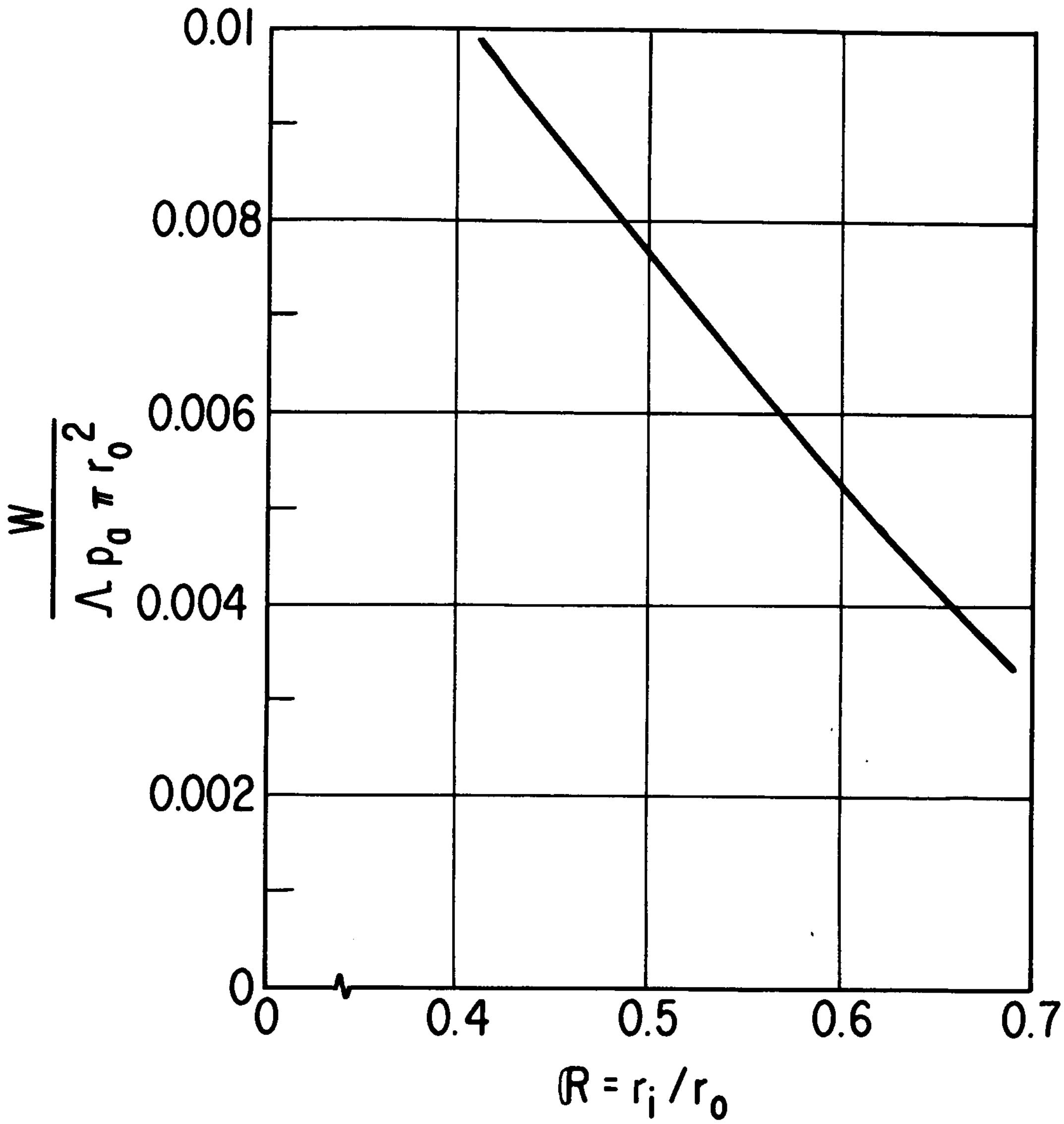

Fig. 9 Load Capacity of Optimized Flat Thrust Bearing.

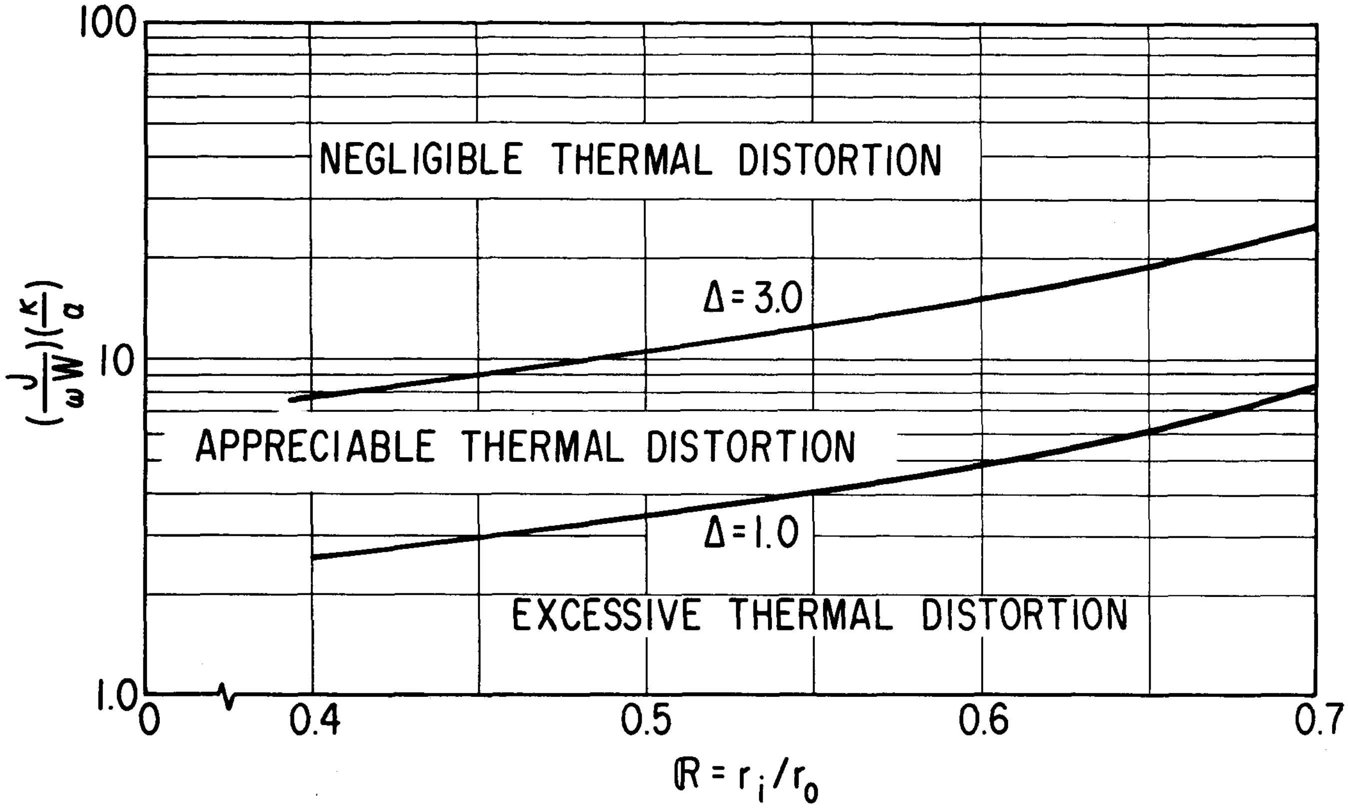

Fig. 10 Thermal Distortion Criteria.

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